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*Lubrication of Non-Metallic Bearings**

By GEORGE E. REISER

Bethlehem Steel Corp., Lackawanna, N. Y.

The art of manufacturing bearings of the phenolic type, has been broadened with the passing of each year, and their use has gradually been extended from their first installations on light loads at relatively high speeds, to heavy loads and moderate speeds. The quality has also been improved to the point where they are adaptable for most conditions in present day applications, and after more than 10 years of extensive use, synthetic bearings have now become definitely established in steel mills. Because their characteristics are entirely different from those bearings of the metallic type, it is necessary for operating men to become familiar with a few simple requirements before they can use them successfully. This discussion, composed of excerpts from previous papers, conveys experiences and other information which should promote a better understanding of the lubrication of composition bearings.

All bearings, whether metallic or composition, are limited in performance by the maximum allowable temperature of the material. All bearings generate frictional heat, the control of which will spell success or failure in any application. Friction is the result of surface contact and is governed by the relative smoothness and hardness of the surfaces involved. The roll neck or journal should be smooth and free from scratches and pits. Where heavy bearing loads are encountered, the finishing of the necks must be done very carefully, because this is one of the important factors in synthetic bearing operation. Good chuck fit is next in importance, and the chucks and the bearings should therefore be machined to a good fit. When the contours of chucks and bearing liners are not in perfect agreement, hot spots will develop due to the concentration of the load, and failure is likely to occur if the condition is not corrected. Assuming that all bearing liners have a perfect backing in a chuck strong enough to resist bending strains, the contact area is the next consideration.

Regardless of make-up, bearings of the phenolic type have a fine resin film on the surface which is hard and slightly abrasive in action. This increases the operating temperature in the running-in period and calls for extra care for the first few hours of operation. Going even further a bearing may have a perfect chuck fit, roll neck in excellent condition, skin removed from the bearing surface, and still operate hot. This behavior can be caused by too large an arc of contact for the bearing. Our experience is, an arc of contact over 120° is not desirable for synthetic bearings due to the resiliency of the material which, under heavy loads, will spring in and produce a choking effect on roll neck or journal. For light bearing loads this behavior need not be considered. In some designs attempts are being made to eliminate this clamping by using a bearing with an inside diameter up to 1/16 inch larger than the outside diameter of the roll neck. This is occasionally worse than the clamping effect itself, because, if the load is not sufficient to compress and cold-flow the material to the form of the roll neck, a heavy load concentration will take place, resulting in hot spots and eventual wiping of the bearing. Bearings that are jaw tight due to changing from roll to roll of different journal diameters will squeal and burn when the mill is first started. Swabbing of bearings and journals in roll changes will prevent this. Thrust collars are bound to wear excessively regardless of lubrication, if they don't fit perfectly into the fillet. If the collar has too small a radius, it will climb the fillet, causing spotty contact and burning. Lubrication at its best cannot rectify such a condition. After a mill has been set up, rolls and bearings changed and pulled up tight, it is always good practice to back off the thrust adjustment screws a flat or two. This relieves the collar sufficiently to allow water to penetrate and lubricate. With synthetic resin bearings, the arc of contact should be just large enough to maintain proper roll alignment, that is, it may be much smaller than for metal bearings. Under light loading, danger from hot spots is not serious. The proper contact area is gradually established in operation, at a point where wear is at a minimum.

As more experience is being gained in operating these bearings, it becomes easier to suggest workable ideas in regard to the arrangements of scale guards, water guards, wipers and other devices that are needed on many installations. Scale guards are necessary on plate mills, blooming mills and other large stands. Different types of guards are used, depending upon whether hot top ingots are rolled, or only the usual quantity of heavy mill scale is encountered. Water guards have been developed which effectively keep water off dry rolling stands. Pinion stands are equipped with still a different type of guard, easily installed on open-type housings.

So far we have considered the reduction of frictional heat, by providing perfect surfaces and bearing fits. However, due to the low heat conductivity of composition bearings, a cooling medium is required for regulating their operating temperature. Knowing the critical temperature and the rate of heat generation, the problem is to deter-

* Courtesy Association Iron and Steel Engineers

mine the rate of cooling, which will permit operation below this temperature. Cooling of roll necks or journals is generally accomplished by the use of water. The amount of water required for this purpose is dependent on several factors.

(a) The heat generated in the bearing itself.

(b) The type of work, whether continuous or intermittent, with light or no load periods.

(c) The amount of heat transmitted from the roll body.

(d) The temperature of the cooling water, its location, and the exposed roll-neck area.

An easy flow of water with ample volume, is usually more effective than a jet-type pressure spray. The water should cover as much of the exposed journal surface as possible. When bearings cover the top and bottom of the journal the water should be applied to both sides between the top and bottom bearings.

A lazy flow of water of good volume washes out the pepper carbon scale and also has a more definite cooling action than a spray. Sprays do not cool as efficiently as the swiftly flowing water will not allow heat transfer readily. Hydraulic sprays are detrimental insofar as they blow the pepper carbon into the bearings. Descaling sprays should always be set away from the mill.

Several precautions must be taken in connection with the water used. First, its temperature is a factor in the rate of heat transfer; second, it must be free from abrasive impurities such as dirt, sand, etc.; and third, it must be free from chemical impurities such as salt. When there is a trace of salt or acid in the cooling water, a surface breakdown of the roll neck is sure to follow, resulting in abnormal wear on the bearing. On the other hand, water in a closed system may be treated to give increased bearing lift. It is not the intention here to enter into the theories and facts behind the last statement. However, it is observed almost daily that under certain load, speed, temperature and water conditions a dark film forms on the journal or the neck surface and that, as soon as this film is in evidence, a reduction in power consumption and a drop in operation temperature follow. The formation of this film is partly due to chemical and partly to mechanical causes. The important fact is, however, that the separation of the bearing surfaces by this film results in a higher efficiency in operation.

This phenomenon prompted the investigation of lubricants as a means of producing a separating film between the rubbing surfaces. Water-soluble oils at once showed their inability to maintain a separating film even under moderate load conditions. The same was noticed for most of the water-repellent oils investigated. The reason for this is the inability of the oil to adhere to the rubbing surfaces. Some of the high-pressure greases operated satisfactorily at speeds of 200-250 f.p.m. and medium loads up to 1750 p.s.i., but failed badly at low speed on high bearing pressure applications. The final investigation was made with extreme pressure quality greases, and here compounds containing tallow or suet proved superior to most other types of extreme pressure media.

We now use grease containing graphite, which we contend "conditions" both the neck and the bearing. The graphite works into the pores of the steel to give it a burnished appearance. After the resin coating is worn away from the bearing surface, the graphite is held in the fabric to give added lubrication. This has been observed time and again after the change of rolls. When the top bearing is raised off the neck, a perfect pattern of the fabric is molded in graphite on the neck. It has also been found that as long as the roll is in storage, a very small amount of rust is found, showing the protective action of the graphite.

Tests on lubricants proved the importance of water cooling to maintain a reasonable operating temperature, as it was noticed that, with the exception of the self-generated dark film referred to, all other grease films gave a higher coefficient of friction. The friction was dependent on the lubricant used and not on the type of the synthetic bearing. Where speed and load conditions do not encourage lubrication of synthetic bearings with water only, the addition of the proper type of grease will permit satisfactory operation and increase bearing life. Each bearing has its own characteristics and a lubricant satisfactory for one will not necessarily do for another. Some experimentation may be required on each application as well as on the different makes of synthetic bearings to select the lubricant best suited for the work. First, a grease should have a film strength in excess of the highest bearing pressure possible in the particular case; it must adhere to the roll neck and resist the wiping action produced under load; it must not be soluble, and it must have a high melting point. It should also be kept in mind that a fine film of grease is much to be preferred to a heavy coating of lubricant. With these points in mind the search for a lubricant is a minor problem and the benefits received will shortly compensate for the small amount of work involved.

Automatic grease systems have proved very satisfactory, especially in blooming mills. A small shot of grease to a bearing, administered periodically, retains a film and maintains economical lubrication. Grooving of fabric bearings is not essential. We have found that a heavy chamfered grease entry port is all that is necessary. This forms a pocket from which the grease spreads effectively.

With proper lubrication and fitting of non-metallic bearings, a saving as high as 35% in power has been realized. To achieve this saving the following suggestions are presented:

1. Roll necks or journals should be polished smooth, free from checks and pits.

2. Chucks or bearing housings should be strong enough to resist the maximum bending force possible on the unit and to fit bearings to a point of suction.

3. Synthetic bearings should be heavyweight fabric.

4. Arc of contact of bearing should be just large enough to maintain roll alignment.

5. Cooling water should be adequate to remove all heat generated in operation and should be free from abrasive and chemical impurities.

6. Grease lubrication, with water as a coolant, is advantageous on all low-speed, high-pressure installations. The grease film should be capable of resisting rupture even under maximum pressure.

Literature Cited: "Composition of Bearing Applications in Industry" by John A. Petho and Frank W. Vogt.

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Lubrication of Eaton Two-Speed Axle

By R. H. GALE, Planning and Order Department, Eaton Manufacturing Company Cleveland, Ohio

Inadequate lubrication is a major contributor to unsatisfactory axle life. All the high quality material and workmanship put into the axle will be entirely offset by poor lubrication. Although inexpensive in itself, neglect in selecting a correct lubricant, or failure to change at proper intervals, always results in expensive repairs. (Inferior oils do not have sufficient film strength and the film strength of good quality oil may be lost by use over too long an interval between changes.)

Suitable oil in the Eaton Two-Speed Axle is even more important in view of the heavier loads the truck—and consequently the gears—are called upon to handle.

LUBRICANT REQUIRED

To insure long gear life, use nothing but the best Extreme Pressure (EP) Lubricant. A list of the makes recommended and approved has been compiled by Eaton engineers. Should you desire a copy of this listing, send your request to Eaton Manufacturing Company, Service Department, Cleveland, Ohio.

The most satisfactory results will be obtained only when the lubricant is of the correct viscosity.

For all normal temperatures use SAE 90 EP. For very extreme high temperatures consistently above 100° F. use SAE 140 EP.

LUBRICANT CHANGES

Frequency of oil changes in the Two-Speed Axle depends upon the type of service in which the vehicle is operating. When in severe heavy work, the oil should be renewed every 5,000 miles, but if the work and loads are light, changes at 10,000 mile intervals will be satisfactory.

At the time of lubricant change, drain the axle completely and flush with light motor oil or flushing oil before refilling. Run the truck without load (or on jacks) in the low range to make sure that flushing oil goes through the planetary gears. After again draining, refill with recommended lubricant.

CHECKING BETWEEN CHANGES

Oil level in the axle should be checked at 1,000-mile intervals. Make-up quantity should be added as needed.

The various producers of Extreme Pressure Lubricants use different methods and chemicals to impart extreme pressure quality to their axle oils. Very often the mixture of two different brands of EP lubricants will result in chemical reaction and the loss of EP quality. *Never add lubricant to axle unless it is the same make and grade as that which is already in the axle.* If the same lubricant is not available, drain, flush, and refill as recommended in these instructions.

FILLING INSTRUCTIONS

Fill axle until oil flows from overflow plug which is located in the axle housing cover. Insert overflow plug, then add one pint of oil, using filler hole at top of carrier. This will provide the extra oil required to fill the differential and planetary unit. Be sure the axle is not overfilled. For routine refill, do not fill above plug level. At initial running-in of chassis, the axle should be left in low axle ratio and run without load for about one mile in order to circulate oil through the differential unit.

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Some Technical Aspects of Launching Lubricants

By CHARLES W. BOHMER, JR., *Standard Oil Company of New Jersey*

(Continued from October Issue)

One point of interest is the abrupt change in the curves for all three pressures at a worked penetration of approximately 250. Thus, based on these data, there would be no advantage in using a grease of the type tested softer than this value.

The variation in consistency of a typical slip coat with temperature is shown in Figure 73. The balance of the data, both from laboratory tests and actual launchings, presented in this paper will be on the same slip coat. Since it was not feasible in these laboratory tests to obtain coefficient of friction data on this grease at various temperatures, the data in Figures 72 and 73 were combined to give variation of coefficient of starting friction with temperature. Such data are given graphically in Figure 74 for a pressure of 1.75 long tons per sq. ft.

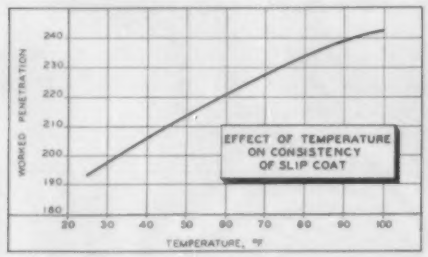

Fig. 73

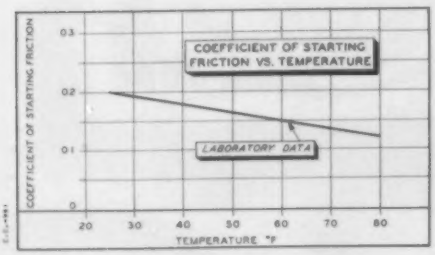

Fig. 74

Data From Actual Launchings

Lack of suitable data from launchings has made it practically impossible to judge how well such data would correlate with actual conditions. Recently some data were made available to the author which are ideal in that they are from the same yard on the same type of ship, using ways all of which had nearly the same declivity, and using the same base coat and slip coat combination applied in the same way and amount by the same crews. The initial unit pressures, which were all within very close limits, were near to the value of 1.75 used in the laboratory data. Consequently, the only known variable was temperature.

Fortunately, grease temperatures underneath the ship were taken. A comparison of grease and air temperatures taken before the launchings are shown in Table 4.

The coefficient of starting frictions calculated from the launching data are plotted in Figure 75. The experimental data are shown for comparison. The values from the launchings are higher than those obtained in the laboratory and they increase at a somewhat faster rate with decrease in temperature. Unfortunately laboratory data are not available below 25° F. so we do not yet know whether they would show the same increase indicated by the launching data at about this temperature. It is hoped that more data can be obtained in the range from 0° to 30° F. both in the laboratory and in launchings to determine more accurately the shape of the curves in this region. It is also hoped that data will be available which will permit an attempt at correlation of starting friction data with different slip coats.

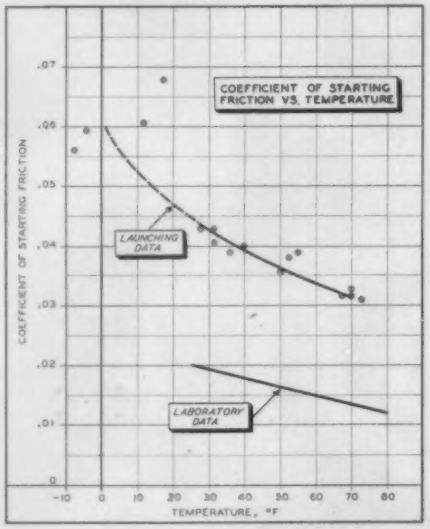

Fig. 75

Plotting Rate of Shear vs. Shearing Stress

Because of the experimental error involved in obtaining coefficient of starting friction data with the apparatus shown in Figure 70 and the difficulty in obtaining such data over a wide temperature range, it was decided to approach the problem of correlation of laboratory and launching starting friction data from more basic grease data. In the paper "Flow Characteristics of Lubricating Greases," presented at the N.L.G.I. meeting last year by Dr. J. C. Zimmer, it is pointed out that the yield value of a grease is the force required to start motion of the grease. It seemed logical to attempt to correlate the yield value of the slip coat over a range of temperatures with the force necessary in the launchings to overcome starting friction.

Accordingly relative yield values at several temperatures were obtained by plotting rate of shear vs. shearing stress and determining the intercept of the axis of abscissas (Figure 76). The values were obtained as dynes per square centimeter and were converted to pounds per square foot as shown in Table 5.

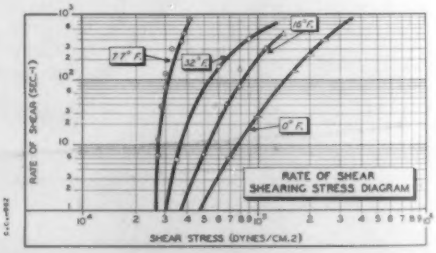

Fig. 76

Table 4

AIR TEMP. °F.	0	20	17	32	36	40	64	62	70	76	70	72	76
GREASE TEMP. °F.	−4	12	18	28	32	36	50	56	63	67	70	70	73

Table 5

TEMPERATURE ° F.	ESTIMATED RELATIVE YIELD VALUES	
	DYNES/SQ. CM.	POUNDS/SQ. FT.
77	26,000	52.7
32	28,000	56.7
16	36,000	73.0
0	48,000	97.2

The force required to start motion in the case of the actual launchings and the laboratory tests were recalculated from the formula for coefficient of starting friction. In this case f and P are known values and F can be calculated in pounds per square foot. These data are plotted in Figure 77. It is obvious that these yield values do not correlate well with either these launchings or laboratory data.

After the ship has started, launching greases have considerable effect on the speed of launching. Figure 80 illustrates the resistance offered by the grease in a typical launching. Since the resistance of the grease can be translated into coefficient of sliding friction, which in turn should be dependent on the grease viscosity, it was thought advisable to see if a correlation could be made between coefficient of sliding friction and grease viscosity.

Many variables have to be considered and assumptions made. Since coefficient of sliding friction varies as the ship travels down the ways, it was decided to use the minimum coefficient of sliding friction corresponding to the maximum acceleration. Figure 79 shows some launching data giving time, velocity, and acceleration curves versus distance traveled. The minimum coefficient of friction is calculated by substituting the value for maximum acceleration obtained from the curve in the equation:

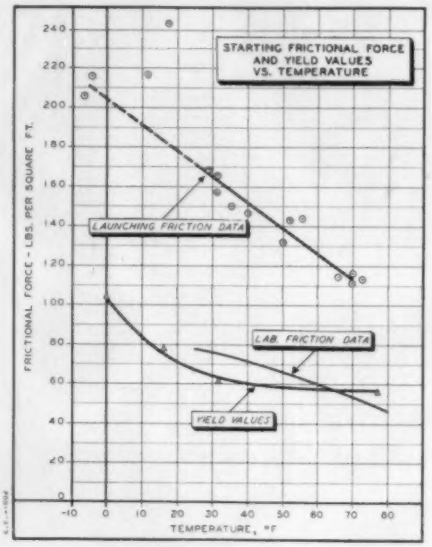
STARTING FRICTIONAL FORCE AND YIELD VALUES VS. TEMPERATURE
FRICTIONAL FORCE - LBS. PER SQUARE FT.
LAUNCHING FRICTION DATA
LAB. FRICTION DATA
YIELD VALUES
TEMPERATURE, °F

Fig. 77

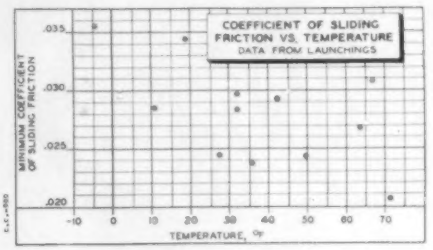
COEFFICIENT OF SLIDING FRICTION VS. TEMPERATURE
DATA FROM LAUNCHINGS
MINIMUM COEFFICIENT OF SLIDING FRICTION
TEMPERATURE, °F

Fig. 78

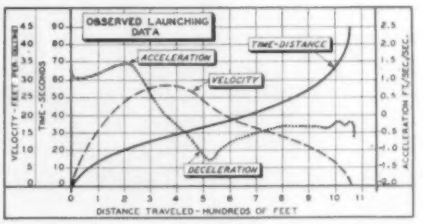

Fig. 79

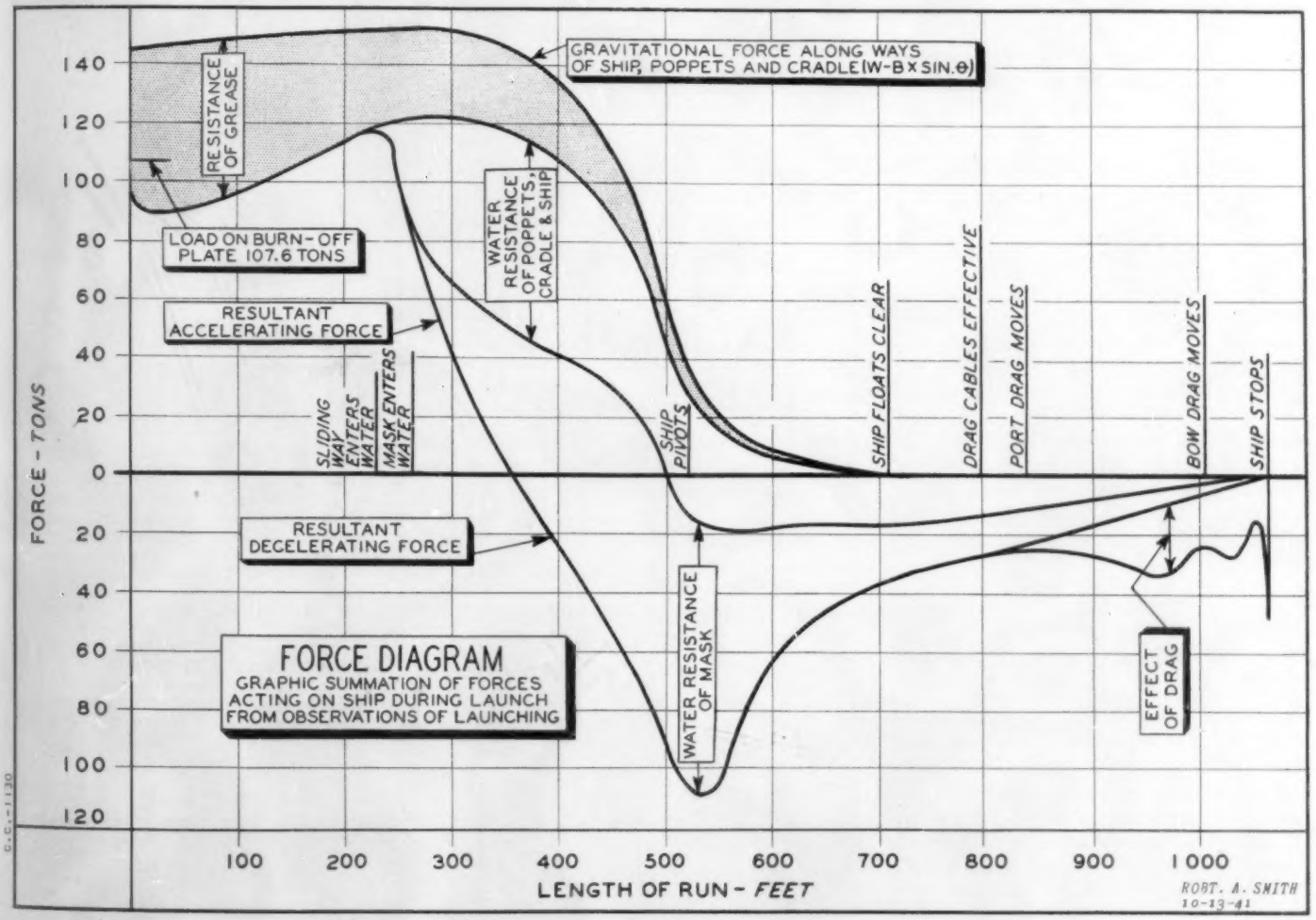
GRAVITATIONAL FORCE ALONG WAYS OF SHIP, POPPETS AND CRADLE (W-B x SIN. θ)
RESISTANCE OF GREASE
LOAD ON BURN-OFF PLATE 107.6 TONS
RESULTANT ACCELERATING FORCE
WATER RESISTANCE OF POPPETS, CRADLE & SHIP
SLIDING WAYS ENTERS WATER
MASK ENTERS WATER
SHIP PIVOTS
SHIP FLOATS CLEAR
DRAG CABLES EFFECTIVE
PORT DRAG MOVES
BOW DRAG MOVES
SHIP STOPS
RESULTANT DECELERATING FORCE
WATER RESISTANCE OF MASK
EFFECT OF DRAG
FORCE DIAGRAM
GRAPHIC SUMMATION OF FORCES ACTING ON SHIP DURING LAUNCH FROM OBSERVATIONS OF LAUNCHING
FORCE - TONS
LENGTH OF RUN - FEET
ROBT. A. SMITH 10-13-41

Fig. 80

$$f = \tan x - \frac{a}{32.2 \cos x}$$

where

f = minimum coefficient of sliding friction.

a = maximum acceleration.

x = angle ways make with the horizontal.

To obtain grease viscosities, rate of shear would have to be known. Under launching conditions this would be calculated by dividing the velocity of the ship by the distance between the base coat and the sliding ways. This is a varying thickness, but a range of 1/20″ to 1/100″ would probably cover most conditions. Assuming a range of velocity of 15 to 30 ft./sec. would give us a range of shear rates of 3,600 to 36,000 reciprocal seconds. By plotting the apparent viscosity-rate of shear curves for the grease at several temperatures and taking values in the range of shear rates involved it was hoped a correlation might be obtained with the minimum coefficient of sliding friction data.

Unfortunately the data on coefficients of sliding friction, obtained from the same launchings as the coefficient of starting friction data previously shown, when plotted against temperature were so scattered (Figure 78) that no correlation could be attempted.

There are various reasons why it may be difficult to obtain correlation of such data. The actual temperature of the slip coat as the ways pass over it at maximum rate of acceleration may well have little or no relation to the starting grease temperature or atmospheric temperature. Varying thicknesses of slip coat would give variable rates of shear which in turn would affect the grease viscosity and the coefficient of sliding friction. There also may be an unknown effect of base coat-slip coat interlayer. The distances traveled before the ships hit the water would also affect the results, but it was hoped that in these launchings the distances were uniform enough to avoid any effect due to this.

Some little work also has been done in trying to correlate coefficient of sliding friction when greases of different types and viscosities were used but the data obtained so far are also inconclusive.

In the beginning of this paper it was stated much of the data presented are of a preliminary and exploratory nature. We repeat this admonition in closing; however, if its presentation will bring out additional data of similar type by other investigators its preparation will have been justified.

(Concluded)

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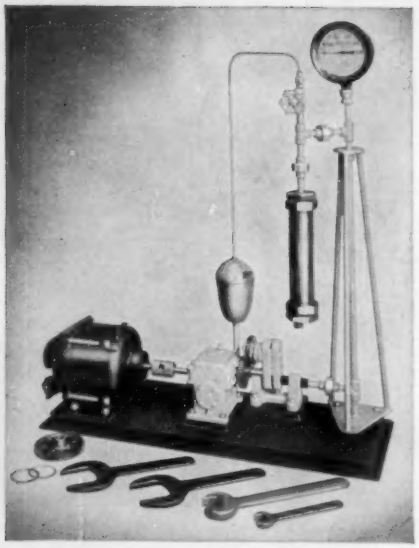

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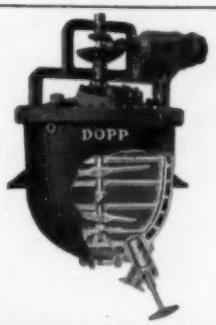

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